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NUT FACTOR STUDIES IN BOLTED JOINTS

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ABSTRACT

In the design of rotating machinery it is often necessary to bolt components together with a known preload since the same controls the system dynamics considerably. Since a major percentage bolt tightening torque is resisted by frictional forces especially towards the end of tightening, and since the coefficient of friction in threads and under the bolt head to a large degree is indeterminate, the best method of ensuring desired clamping load is by measurement of bolt elongation, or by a load cell. However in large number of cases the incorporation of load cell is not practical and bolt elongation is insufficient to allow accurate measurement. In such cases it becomes necessary to produce the desired end load by applying a specified torque to the bolt leading to the determination of nut factor for each of the cases

Here an attempt has been made to obtain the quantitative values of nut factors for bolted joints as a function of bolt diameter and tightening torque applied on bolted joints. A simple test rig was designed and fabricated for the purpose in which the experiments were conducted. The instrumentation basically includes the monitoring of the bolt tightening torque through the torque wrench, and a load cell which measures the clamping force developed in the joint. These experiments were conducted with thrust ball bearing and also without thrust ball bearing; resulting in possible analysis of the friction generated between nut and bolt threads and also between nut-washer surfaces. The results include generation of nut factors and coefficient of friction values developed for different torques and different bolt diameters, by using a mathematical relation which relates the bolt geometry, clamping force and the tightening torque applied. Also an effort has been made to develop a curve fit equation for nut factor values from experimental data leading to quantitative estimation of nut factors as a function of bolt diameter and tightening torque.

KEY WORDS : Bolted Joint, Nut Factor, Torque, Clamping force

1 INTRODUCTION

The clamping force on joint is initially created when the components are assembled using a bolted joint, which also creates tension in the bolt. The reaction to the tension is usually called preload in this stage although there may be some plastic deformation in some of threads when the bolt is tightened with an initial torque [1]. Most of the bolt and joint members respond elastically as the bolt is tightened. The joint members are compressed to a slight amount, and the bolt is stretched by larger amount [2].

The clamping force, a bolt exerts on joint is usually called preload in the bolt. In other words, we apply a torque on the nut, which leads to the turning of nut, the stretching of bolt and generation of preload in the bolts. The easiest and least expensive way to control preload is by torque or turn, because these are the inputs to the system.

The concept of preload control and Nut Factor

Preload in bolted joints should be as high as components can withstand to move towards a rigid joint. While a part of the tightening torque applied on nut results in generating the preload, the rest gets dissipated in overcoming the thread friction, and nut-bearing friction [3]. The preload results from what is left of the tightening torque after friction has been overcome. Anticipated variations in the coefficient of friction will thus have a marked effect upon the preload, and careful calibration of the torque wrench will do little to improve accuracy. Each application should be tested to determine its own nut factor since the nut factor includes the effects of everything that affects the torque to preload transformation, not just friction. Also the consequences of local deformations, interrupted tightening, inaccurate thread form, misaligned component, relaxation due to tightening other bolts in the joints, whether the fastener is new or is being reused, initially bent bolts and speed of tightening etc, all of which affects the preload reproductively.

The nut factor K can only be determined experimentally, and experience shows that the same should be determined for each new application so that the range of pre-load can be predicted with reasonable level of confidence.

Torque – preload equation is given by [3],

$$T=F_p.*(K*D) \qquad (1.1)$$

Where K is known as nut factor that is an experimental constant used to evaluate or describe the ratio between the torque applied to a fastener and the preload achieved as result.

$$K= T/(F_p*D). \qquad (1.2)$$

Tightening torque – clamping force relationship in a bolted joint

Whenever a torque is applied on nut of a bolted joint, the nut moves in axial direction along with rotation about bolt axis. This induces a clamping force in a bolted joint. In any bolted joint, the total tightening torque on nut has to overcome two frictional resistances; one is bolt thread frictional resistance and other is the nut-bearing (or washer) surface frictional resistance.

The torque T_1 required to overcome the bolt thread frictional resistance is given by the relation [4],

$$T_1 = Fp.\frac{d}{2}\left(\frac{cos\beta.tan\alpha + \mu_1}{cos\beta - \mu_1.tan\alpha}\right) \qquad (1.3)$$

and

$$\tan\alpha = \frac{1}{\pi p d}$$

Where
d = pitch diameter of bolt threads, m
Fp= Preload induced in the bolt, N
â = one half of thread angle, degrees
á = Thread lead angle, degrees
ì_t= Coefficient of friction in threads.
p= Number of threads per m

In case of bearings which are newly fixed in position, the intensity of pressure is assumed to be uniform over the surface, but since points at different radii are moving at different speeds the coefficient of friction may not be constant at all radii. Also when motion takes place, the rate of wear will be greater at those parts where the velocity of rubbing is greater, that is outer radii. This will simultaneously alter the distribution of uniform pressure. The pressure will tend to increase from outer radii to center. This alteration of pressure again alters the rate of wearing. Hence it is difficult to estimate how bearing may wear or how the coefficient of friction may vary.

In order to overcome the above difficulty, one of the following assumptions is made though neither of them is strictly true. Those assumptions are (a) the pressure is uniform and (b) the wear is uniform. In uniform pressure theory, the intensity of pressure is assumed to be acting uniformly at all the points on nut bearing surface. If the nut used is a new one, this theory can be applied. The surface coefficient of friction is assumed to be uniform and there is no wear between the sliding surfaces. The velocity of rubbing surfaces does not change the amount of wear considerably between the new surfaces. Therefore for new nuts used, uniform pressure theory can be applied to find out the required tightening torque, which is given by

$$T_2 = \frac{2}{3}\mu_p \left[\frac{R^3 - r^3}{R^2 - r^2}\right] \tag{1.4}$$

Where
R = Outer radius of nut-bearing surface, m.
r = Inner radius of nut-bearing surface, m.
μ_p = Friction coefficient between rubbing surfaces.

In uniform wear theory, the product of intensity of pressure and velocity of rubbing at every point may be assumed constant, or since the velocity is proportional to radius, the product of pressure and radius may be assumed constant. This theory is suitable for finding out torque required to overcome the nut – bearing surface friction.

And the tightening torque required for overcoming nut-bearing surface frictional resistance is given by uniform wear theory of clutch (Applicable to old nut with considerable wear) as,

$$T_2 = \frac{1}{4} Fp.\mu_2(Do + Di) \tag{1.5}$$

Therefore the effective radius at which the frictional force may be considered to act is

$$r_e = \frac{1}{2}(R + r) \tag{1.6}$$

where

μ_2 = Coefficient of friction under nut bearing surface.

Do=Outer diameter of nut bearing surface, m

Di=Inner diameter of nut bearing surface, m

R = Outer radius of nut-bearing surface, m

r= Inner radius of nut-bearing surface, m

As such the total tightening torque T required to overcome the friction is given by,

$$T = T_1 + T_2$$

Generally uniform wear theory is considered for most of the cases to find out the total tightening torque.

Nut factor can be derived from tightening torque equation as,

$$K = C_1\left(\frac{\cos\beta.\tan\alpha + \mu_1}{\cos\beta - \mu_1.\tan\alpha}\right) + C_2\mu_2 \tag{1.7}$$

where

$$C_1 = \frac{d}{2D}, \qquad C_2 = \frac{Do + Di}{4D}$$

2 METHODOLOGY AND EXPERIMENTATION

A simple test rig was built to obtain torque-clamping force relationships in a bolted joint. The photograph of test rig shown in figure1 consists of a bolted joint, which clamps two flat plates with a load cell introduced in between the plates. The bolted joint is rigidly fixed to the rig, so that only the nut would rotate under tightening torque.

The instrumentation basically includes the use of a torque wrench to monitor the tightening torque applied on the bolted joint, and a load cell which measures the clamping force developed in the joint. Provision was also made to incorporate thrust bearings in place of washers leading to determination of torque expended to overcome the friction between threads and nut-washer surfaces.

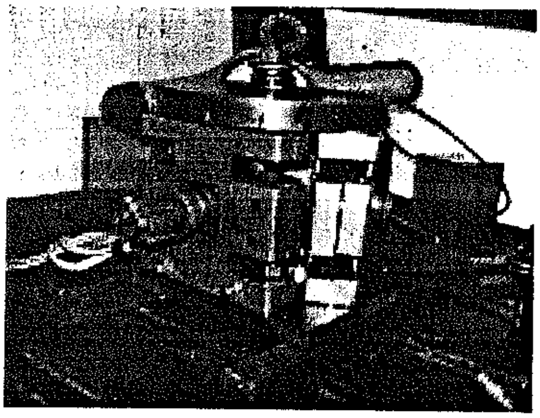

Fig.1. Bolted joint test rig with a load cell

Method of evaluating the nut factors and coefficient of frictions

Applying a known tightening torques on the bolted joints, diffJon, experiments were conducted subjecting bolts of different diameters to varying tightening torques and measuring the resulting clamping load. The results are presented graphically.

Figure 2 shows the results from the experiments conducted on bolts of diameter 8 mm, 10 mm and 12 mm. The results indicate that,

- The clamping force generated in case of the bolted joint without bearing is lesser compared to that of joint with bearing. This is because more amount of friction has to be overcome in case of the joints without bearing. The nut – bearing surface friction has to be overcome resulting in additional requirement of tightening torque.
- As the bolt diameter increases, the clamping force generated in the bolted joint decreases. That is, we need higher tightening torque to achieve more clamping forces as the bolt diameter increases. It is because for the same torque with increasing radius would result in lesser tangential force. This lesser tangential force reduces the magnitude of clamping force generated in the bolt.

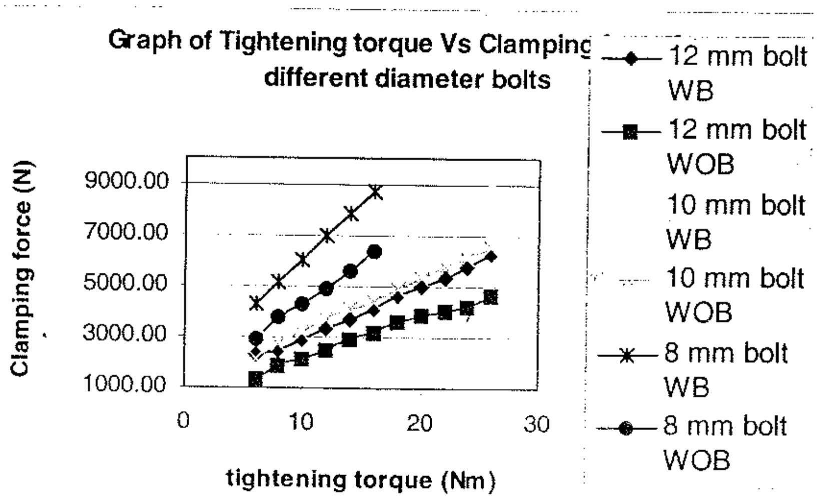

Fig.2 Graph showing the clamping force induced in the bolted joints because of tightening.

Figure 3 indicates the variation of nut factor as a function of tightening torque for different bolt diameters. While it is very clear that the presence of thrust bearings (meaning that the nut washer friction could be neglected) results in relatively lower nut factor, higher bolt diameter yields higher nut factor. The other aspect to be noticed is that the nut factor increases almost linearly with the increase in tightening torque meaning that the rate at which the preload increases, decreases with the increase in tightening torque.

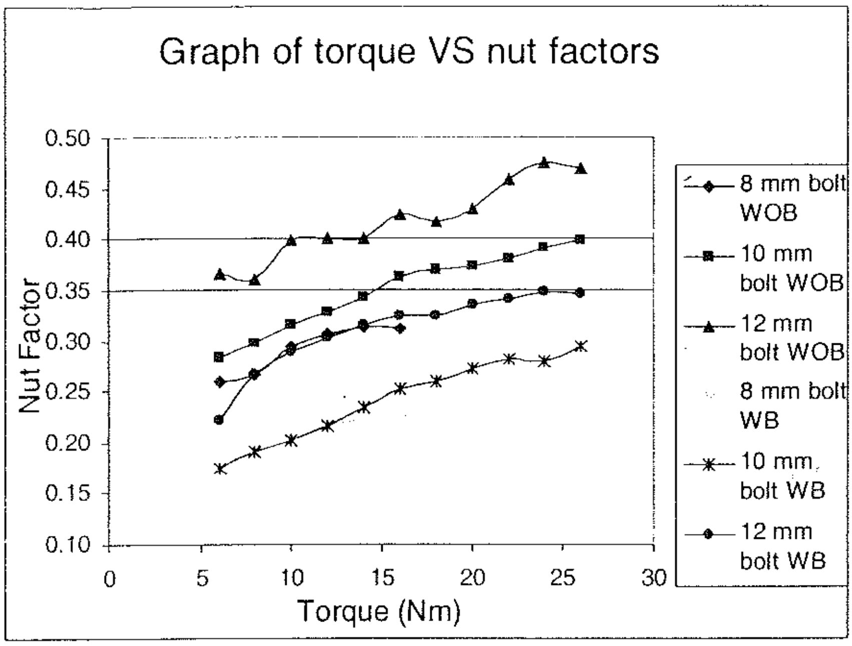

Fig.3. Plot depicting Torque-Nut factors for different bolt diameters.

Determination of coefficient of frictions for threads and nut- bearing surfaces

Figure 4 shows the coefficient of frictions plotted for different bolt diameters which indicate thread and washer friction coefficients. The thread friction coefficient increases for increasing tightening torques. But the washer i.e., nut – bearing surface friction coefficient appears to remain constant with increasing torques.

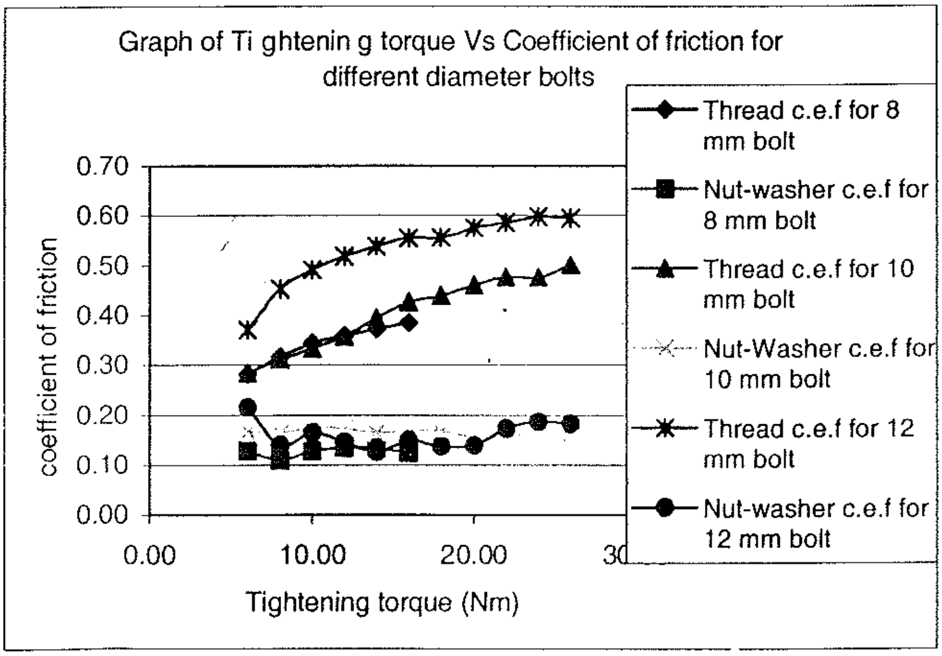

Fig.4 Plot depicting friction coefficients w.r.t increasing Tightening torques

Figure 5 shows different nut factor curves obtained from experiments. Assuming a linear variation for the scattering nut factor values, equations are developed for each diameter bolt which can be written in the form of

$$K = a(T) + b \qquad (1.8)$$

Where K = Nut factor, T = Tightening torque, a, b = constants.

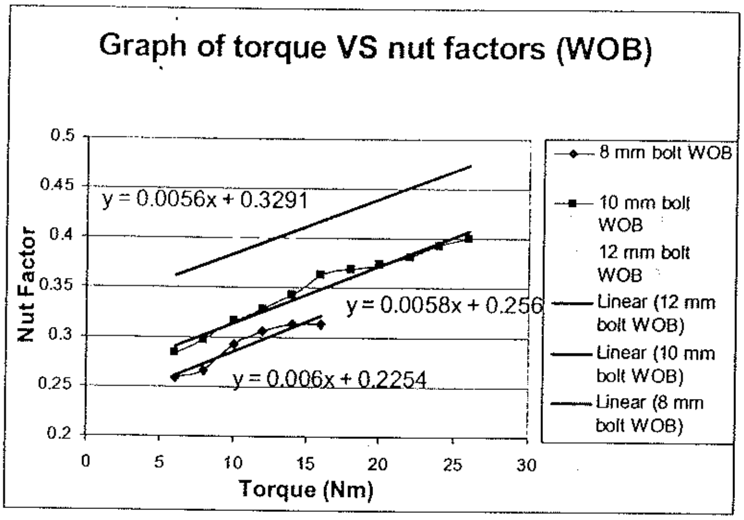

Fig.5 Plot depicting curve fit equations for different bolt diameters

The linear curve fit using the experimental data culminates in the following.

$K = 0.0060(T) + 0.2254$ for 8 mm bolt diameter.

$K = 0.0058(T) + 0.2567$ for 10 mm bolt diameter.

$K = 0.0056(T) + 0.3291$ for 12 mm bolt diameter.

Further the constants in above three equations are assumed to be linearly varying for different bolt diameters and the linear fit is obtained between bolt diameters and constant values as shown below in figure 6.

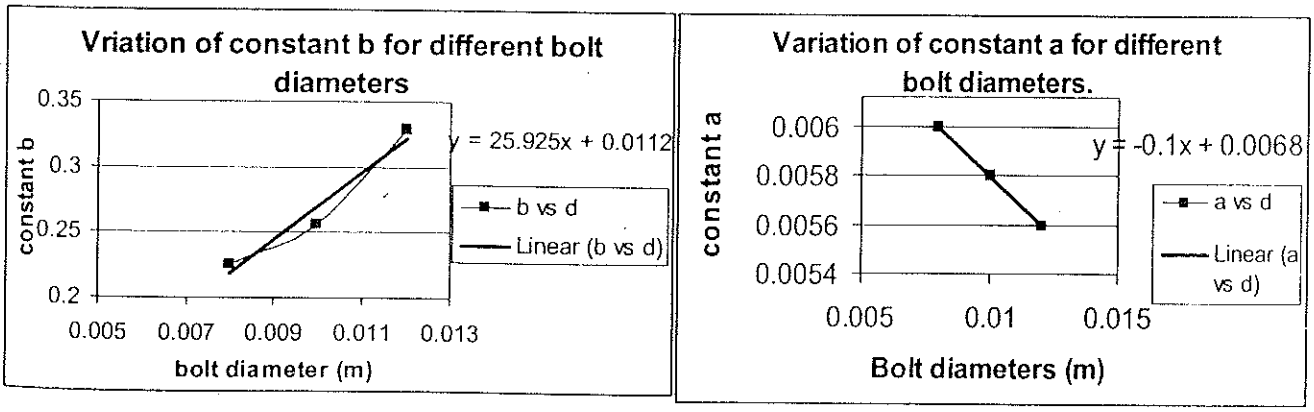

Fig.6 Curve fit equations for variations of constants a,b with respect to bolt diameter

The linear curve fit yields the following expressions from which it would be possible to determine the variation of constants with respect to bolt diameters (expression 1.9 below). Substituting these constants in equations 1.8, we can determine the nut factor for any tightening torque value and bolt diameters (expression 1.10).

$$a = -0.1(d) + 0.0068$$
$$b = 25.925(d) + 0.0112 \quad (1.9)$$
$$K = a.T + b$$
$$K = (-0.1d + 0.0068)T + (25.925d + 0.0112) \quad (1.10)$$

The same is graphically shown in figure 7.

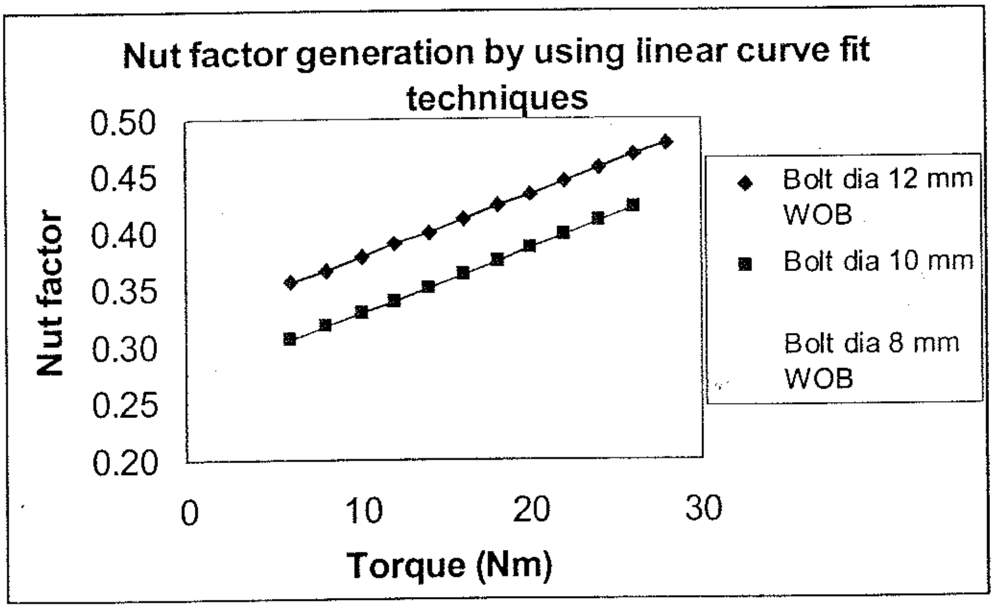

Fig.7 Nut factors generated using curve fit equations.

By using above data we can observe (Table 1) that the nut factor range obtained by curve fit techniques is almost similar to experimental results.

Table 1 Nut Factor

Bolt diameters	Experimental Values of nut factors	Nut factor values by Curve fit techniques
8 mm	0.26 – 0.31	0.25 – 0.31
10 mm	0.28 – 0.40	0.31 - 0.43
12 mm	0.37 – 0.47	0.36 – 0.48

4 CONCLUSIONS

As bolted joints play an extremely important role in controlling the systems' dynamics, it becomes essential to estimate the stiffness in such bolted joints. This necessitates the quantitative estimation of clamping load and coefficient of friction as a function of different related parameters like tightening torque and diameter and hence a need for evaluating nut factors. As such an attempt has been made here to evaluate nut factors in bolted joints with reasonable degree of confidence level.

A simple test rig was developed and fabricated for purpose of experimentation which essentially consisted of bolted joint with facility to apply torque and measure the clamping loads. Thrust bearings were used in place of washer to overcome washer friction coefficient leading to evaluation of coefficient of friction in the threads.

Experimental data has been used to obtain a curve fit relating the nut factor as function of bolt diameter and tightening torque from which the clamping load could be evaluated. This study also throws light on a methodology for evaluating coefficient of friction in bolted joints.

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